11. Note that PIPE-FLO® doesn't rename the pump after a commercially available one has been chosen since it is assuming the user is entering names for all the components. The way to tell if the pump has actually changed from the sizing pump to the commercially available one is by the calculated results shown. If power and efficiency are listed, the user knows that the commercially available pump is being listed in the problem since these values aren't given for a sizing pump.
12. Summary: The pump designation and its major performance parameters are:
    - **a.** Operating point: 100 gal/min flow rate at 218 ft of total head
    - **b.** Pump: 1 ½ × 1 × 8 centrifugal pump; ESP-type; Pump curve ABC1055-1
    - **c.** Pump speed: 3500 rpm   Impeller diameter: 7.125 in
    - **d.** BEP: 61.1 percent
    - **e.** Efficiency at operating point: 58.7 percent (within 96 percent of BEP and to the left of BEP)
    - **f.** The software chose a plastic pipe size for the system to be 2½-in Schedule 40 for both the suction and discharge pipes based on the desired flow velocity of 8.0 ft/s. The actual flow velocity is 6.70 ft/s.
    - **g.** A gradual reducer is required at the pump inlet from 2½-in to 1½-in sizes, and a gradual enlarger is required at the pump outlet from 1-in to 2½-in sizes.
    - **h.** The $NPSH_A$ for the suction inlet to the pump is computed to be 29.06 ft.
    - **i.** The value for $NPSH_R$ for the pump and additional operational data for the pump can be accessed from the software.

13.15 ALTERNATE SYSTEM OPERATING MODES

In Sections 13.9 to 13.14, the focus was on the design and analysis of pumped fluid delivery systems that employed a single flow path and that operated at one fixed condition of flow rate, pressures, and elevations. Important principles of system operation were discussed such as the performance of centrifugal pumps, system resistance curves, the operating point of a pump in a given system, *NPSH*, efficiency, and power required to operate the pump. These fundamentals form the basis for understanding how a fluid flow system works.

Many alternate modes of system operation are in frequent use in a wide variety of industrial applications that build on those fundamentals, but that include additional features and that require different methods of analysis. This section will describe the following:

- Use of control valves to enable system operators to adjust the system's behavior to meet varying needs, either manually or automatically
- Variable speed drives that permit continuous variation of flow rates to fine tune system operation and to match levels of delivery to product or process needs
- Effect of fluid viscosity on pump performance
- Operating pumps in parallel
- Operating pumps in series
- Multistage pumps

Reference 7 and Internet resource 2 offer more extensive treatment of these topics beyond what is practical for inclusion in this book, and in a manner that is highly compatible with the terminology and analysis methods presented here. Several other references offer additional coverage as well, particularly References 3–6 and 9–20. Reference 8 is a source for an extensive set of fluid property data (viscosity, density, specific gravity, vapor pressure), steam data, friction losses in valves and fittings, steel and cast iron pipe data, electrical wiring, motors, and controls. Other references relevant to these topics appear in the References section for Chapter 11.

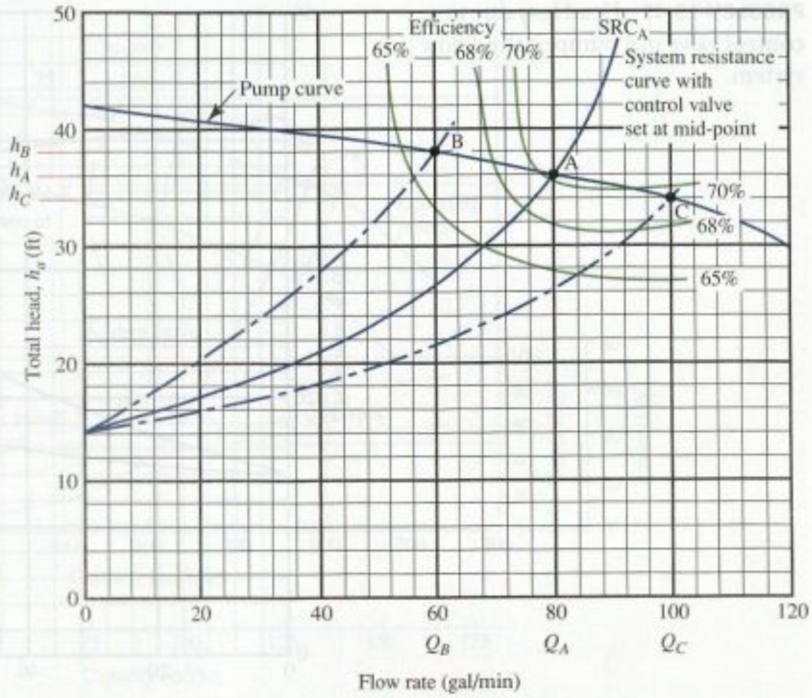
13.15.1 Use of Control Valves

It was stated in Section 13.13 that the operating point of a pump is defined as the volume flow rate it will deliver when installed in a given system and working against a particular total head. The piping system typically includes several elements described in previous sections on the design of suction and discharge lines; valves, elbows, process elements, and connecting straight lengths of pipe. Valves were placed in the system to allow the lines to be shut off when performing service or when the system is shut down; thus, they are often called *shut-off valves*. They were typically low-resistance types such as gate valves or butterfly valves and modeled in their fully open position as part of the SRC.

However, when there is a need for varying flow rates to meet different needs, control valves are used that can be adjusted either manually or automatically. Initial sizing of a control valve is often based on the mid-point between the high and low flow rate limits expected in the application. Then the valve can be adjusted to a more open position (less resistance) or more closed position (more resistance) to produce higher or lower flow rates, respectively.

It is important to obtain data from the supplier for control valve performance across its entire range, typically in

PROBLEM 13.46 Operating points for a system containing a control valve at varying control valve settings



terms of the flow coefficient, C_v, as defined in Chapter 10. In U.S. Customary System units with Q in gal/min and pressure in psi, the definition of flow coefficient is:

$$C_v = \frac{Q}{\sqrt{\Delta p/\text{sg}}}$$

The basis for the flow coefficient is that a valve having a flow coefficient of 1.0 will pass 1.0 gal/min of water at 1.0-psi pressure drop across the valve. Alternate forms of this equation are useful:

$$Q = \text{Flow in gal/min} = C_v\sqrt{\Delta p/\text{sg}}$$
$$\Delta p = \text{sg}(Q/C_v)^2$$

When working in metric units, an alternate form of the flow coefficient is used and it is called K_v instead of C_v. It is defined as the amount of water in m^3/h at a pressure drop of one bar across the valve. Use the following equation for conversion between C_v and K_v:

$$C_v = 1.156\,K_v$$

Now, with a control valve (set at its mid-point) in the system along with all other elements, the modeling of the SRC can be done, and a suitable pump can be selected for the operating point *A* as shown in Fig. 13.46. The flow rate at the operating point is the desired nominal flow rate for the system and the resulting total head on the pump can be read from the chart. For the sample data in Fig. 13.46, we read $Q = 80$ gal/min and $h_a = 36.0$ ft.

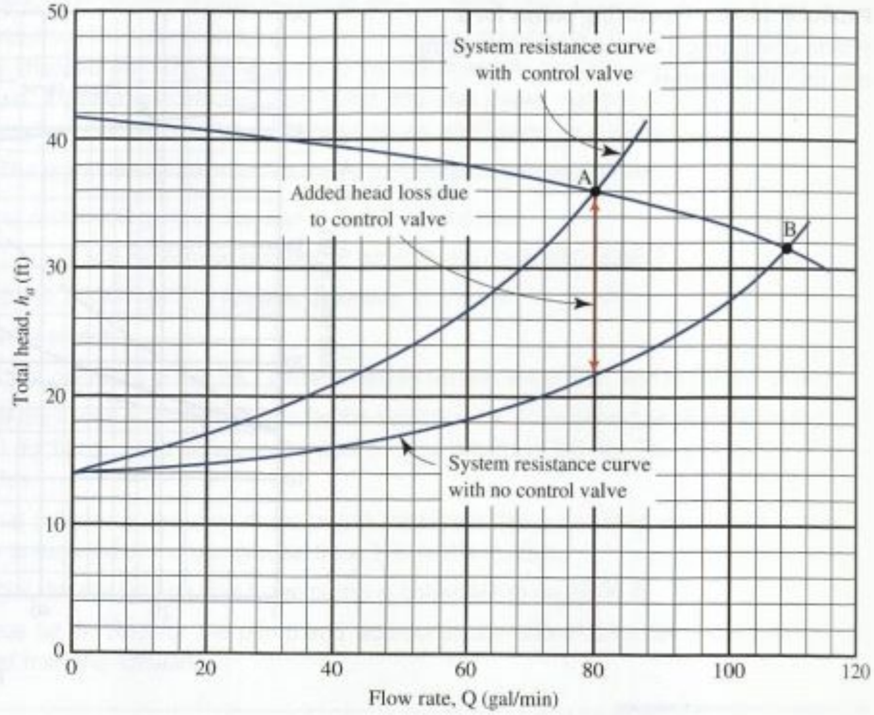
Let's explore what would need to be done if the system operator desired a flow rate of 60 gal/min instead of 80 gal/min. The control valve would be turned to a more restrictive position, placing more resistance to the flow through the system. Then the pressure drop across the control valve would have increased, with a corresponding decrease in the flow and an increase in the total head on the pump. The result is that the SRC would pivot toward the left, reaching a new operating point *B*. At that point, the total head on the pump is 38.2 ft and an additional 2.2 ft of head will be dissipated from the control valve.

If the production system requires a greater flow rate, say 100 gal/min, the control valve will be opened to provide less resistance and the system curve pivots to the right to operating point *C*. At this point, the total head on the pump is 33.5 ft or 2.5 ft less than at point *A*.

It is important to note that other aspects of the pump operation are affected by changing the control valve setting. Figure 13.46 shows pump efficiency curves in the vicinity of the operating points discussed above. The initial operating point *A* results in the pump operating at about 70 percent efficiency, very near the BEP for this pump. When operating at *C*, the efficiency drops to about 68 percent, and at *B* it is about 66 percent. The range of flow rates from 60 gal/min to 100 gal/min is approximately the limit of the range recommended in Hydraulic Institute standards, between 70 percent and 120 percent of the flow at the BEP.

The operation of the control valve inherently involves the dissipation of energy from the system—energy that must be provided by the pump. Therefore, a cost is incurred to perform the control function. Figure 13.47 illustrates the nature of the energy used by the control valve. Curve A is the same as that shown in Fig. 13.46 for the system that includes a control valve set at its mid-point. Curve D is for the same system, but without the control valve. The difference in total head between these two curves represents the additional energy required to perform the control function and the cost for that energy can be calculated. Reference 7 contains extensive discussion about the types of control valves, sizing them to the needs of a particular system, and the costs incurred in their operation.

PROBLEM 13.47 Head loss due to a control valve in a pumped fluid flow system.



13.15.2 Variable-Speed Drives

Variable-speed drives offer an attractive alternative to use of a control valve. Several types of mechanical variable-speed drives and a variable-frequency electronic control for a standard AC electric motor are available. The standard frequency for AC power in the United States and many other countries is 60 hertz (Hz), or 60 cycles per second. In Europe and some other countries, 50 Hz is standard. Because the speed of an AC motor is directly proportional to the frequency of the AC current, varying the frequency causes the motor speed to vary. Because of the affinity laws, as the motor speed decreases, the capacity of the pump decreases; this allows the pump to operate at the desired delivery without use of a control valve and the attendant energy loss across the valve. Further benefit is obtained because the power required by the pump decreases in proportion to the speed reduction ratio cubed. Of course, the variable-speed drive is more expensive than a standard motor alone, and the overall economics of the system with time should be evaluated. See References 7 and 10.

The effect of implementing a variable-speed drive for a system with a centrifugal pump depends on the nature of the system curve as shown in Fig. 13.48. Part (a) shows a system curve that includes only friction losses. The system curve in part (b) includes a substantial static head comprising an elevation change and pressure change from the source to the destination. When only friction losses occur, the variation in pump performance tends to follow the constant efficiency curves, indicating that the affinity laws discussed in Section 13.7 closely apply. Flow rate changes in proportion to speed change; head changes as the square of the speed change; and power changes as the cube of the speed.

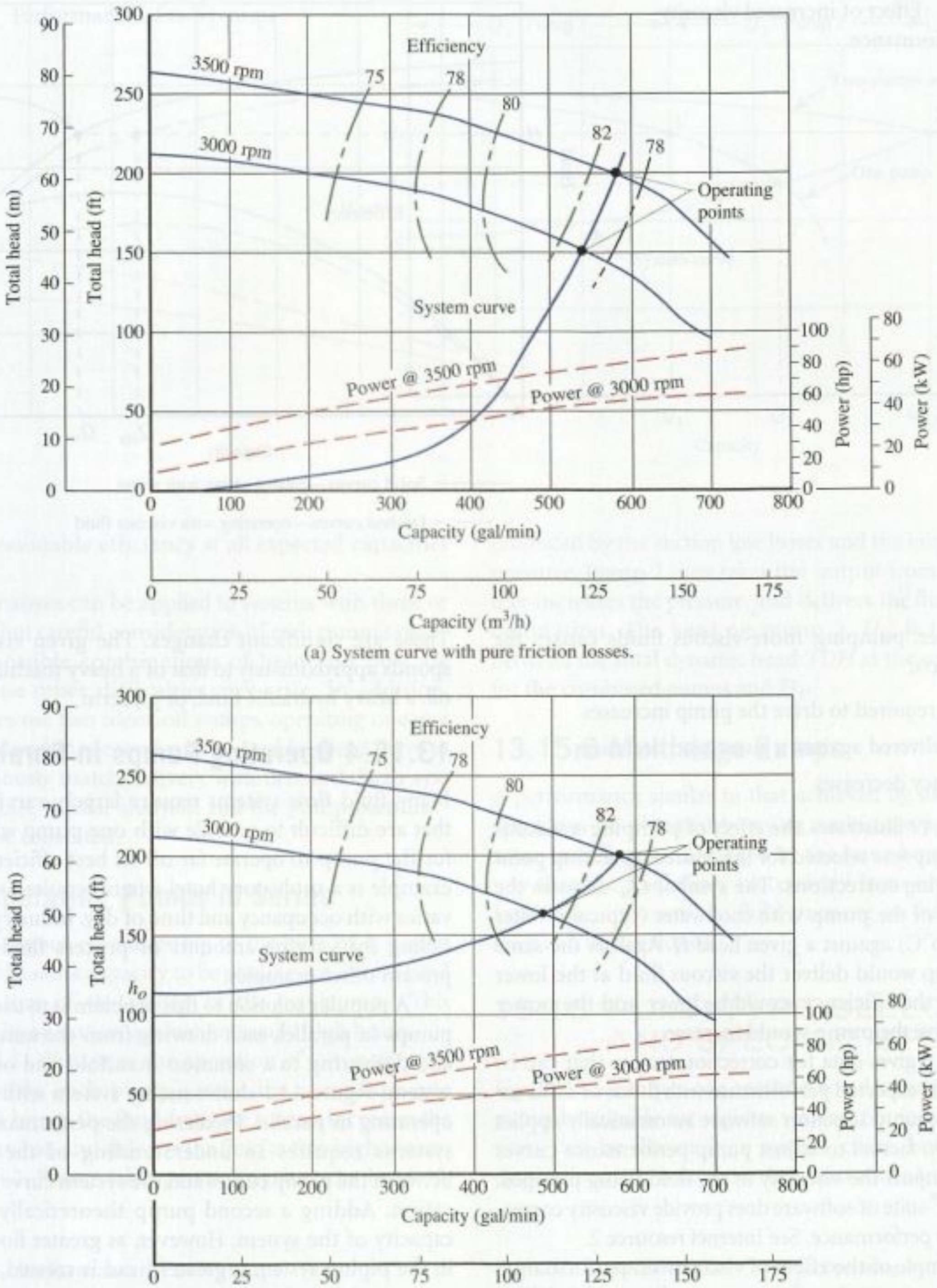
For the system curve having a high static head [Fig. 13.48(b)], the pump performance curve will move into different efficiency zones of operation, so the affinity law on power required will not strictly apply. However, the use of variable-speed drives for centrifugal pumps will always provide the lowest-energy method of varying pump delivery.

In addition to energy savings, other benefits result from using variable-speed drives:

- ***Improved process control*** Pump delivery can be matched closely to requirements, resulting in improved product quality.
- ***Control of rate of change*** Variable-speed drives control not only the final speed, but also the rate of change of speed, reducing pressure surges.
- ***Reduced wear*** Lower speeds dramatically reduce forces on seals and bearings, resulting in longer life and greater reliability of the pumping system.

Operating pumps over a wide range of speeds can produce undesirable effects as well. Moving fluids set up flow-induced vibrations that change with fluid velocity. Resonances can occur in the pump itself, the pump mounting structure, the piping support system, and in connected equipment. Monitoring of system operation over the complete expected range of speeds is required to identify such conditions. Often the resonances can be overcome by using vibration dampers, isolators, or different pipe supports.

The effects of lower or higher flow on fluid system components should also be checked. Check valves require a certain minimum flow to ensure full opening and secure closing of the internal valve components. Solids in slurries may tend to settle out and collect in undesirable regions of the system



(a) System curve with pure friction losses.

(b) System curve with high static head.

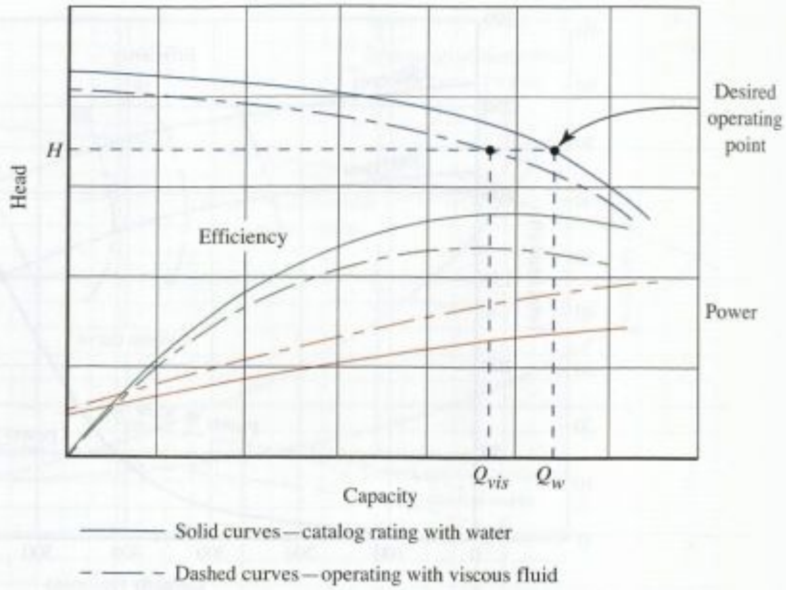
FIGURE 13.48 Effects of speed changes on pump performance as a function of the type of system curve.

at low velocities. Operating pumps and drives at lower speeds may impair their lubrication or cooling, requiring supplemental systems. Speeds higher than normal may require a higher power than the prime mover is capable of delivering, and greater loads are placed on couplings and other drive components.

13.15.3 Effect of Fluid Viscosity

The performance rating curves for centrifugal pumps, such as those shown in Figs. 13.28–13.36, are generated from test data using water as the fluid. These curves are reasonably accurate for any fluid with a viscosity similar to that of

FIGURE 13.49 Effect of increased viscosity on pump performance.



water. However, pumping more-viscous fluids causes the following effects:

- The power required to drive the pump increases
- The flow delivered against a given head decreases
- The efficiency decreases

Figure 13.49 illustrates the effect of pumping a viscous fluid if the pump was selected for the desired operating point without applying corrections. The symbol Q_w denotes the rated capacity of the pump with cool water (typically water at 60°F or 15.6°C) against a given head H. Against the same head, the pump would deliver the viscous fluid at the lower flow rate Q_{vis}; the efficiency would be lower and the power required to drive the pump would increase.

Reference 8 gives data for correction factors that can be used to compute expected performance with fluids of different viscosity. Some pump selection software automatically applies these correction factors to adjust pump performance curves after the user inputs the viscosity of the fluid being pumped. The PIPE-FLO® suite of software does provide viscosity corrections for pump performance. See Internet resource 2.

As an example of the effect of viscosity on performance, one set of data were analyzed for a pump that would deliver 750 gal/min of cool water at a head of 100 ft with an efficiency of 82 percent and a power requirement of 23 hp. If the fluid being pumped had a kinematic viscosity of approximately $2.33 \times 10^{-3}\,\text{ft}^2/\text{s}$ ($2.16 \times 10^{-4}\,\text{m}^2/\text{s}$; 1000 SUS), the following performance would be predicted:

1. At 100 ft of head, the delivery of the pump would be reduced to 600 gal/min.
2. To obtain 750 gal/min of flow, the head capability of the pump would be reduced to 88 ft.
3. At 88 ft of head and 750 gal/min of flow, the pump efficiency would be 51 percent and the power required would be 30 hp.

These are significant changes. The given viscosity corresponds approximately to that of a heavy machine lubricating oil, a heavy hydraulic fluid, or glycerin.

13.15.4 Operating Pumps in Parallel

Many fluid flow systems require largely varying flow rates that are difficult to provide with one pump without calling for the pump to operate far off its best efficiency point. An example is a multistory hotel where required water delivery varies with occupancy and time of day. Industry applications calling for varying amounts of process fluids or coolants present other examples.

A popular solution to this problem is to use two or more pumps in parallel, each drawing from the same inlet source and delivering to a common manifold and on to the total system. Figure 13.1 shows such a system with three pumps operating in parallel. Predicting the performance of parallel systems requires an understanding of the relationship between the pump curves and the system curve for the application. Adding a second pump theoretically doubles the capacity of the system. However, as greater flow rate occurs in the piping system, a greater head is created, causing each pump to deliver less flow.

Figure 13.50 illustrates this point. Observe that pump 1 operates on the lower performance curve and that at a head of H_1 it delivers a flow rate Q_1, which is near its maximum practical capacity at operating point 1. If greater flow is needed, a second, identical pump is activated and the flow increases. But the energy losses due to friction and minor losses continue to increase as well, as indicated by the system curve, eventually reaching operating point 2 and delivering the total flow Q_2 against the head H_2. However, pump 1 experiences the higher head and its delivery falls back to Q_1'. After the new equilibrium condition is reached, pump 1 and pump 2 deliver equal flows, each one half of the total flow. The pumps should be selected so

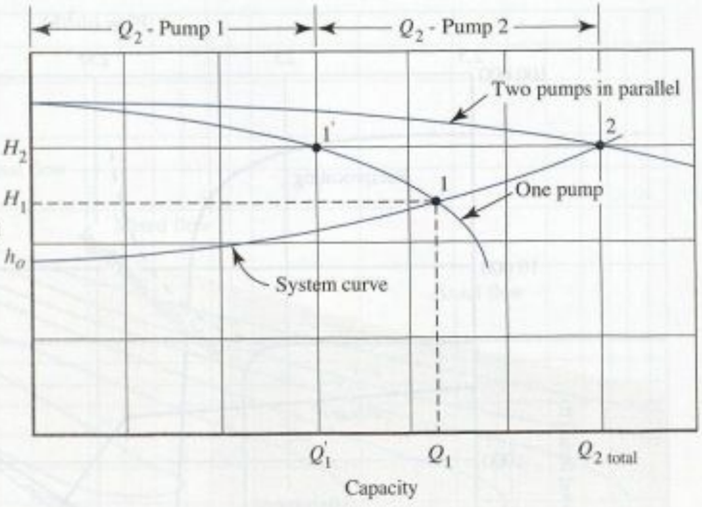


FIGURE 13.50 Performance of two pumps in parallel.

they have a reasonable efficiency at all expected capacities and heads.

Similar analyses can be applied to systems with three or more pumps, but careful consideration of each pump's operation at all possible combinations of head and flow are needed because other difficulties may arise. In addition, some designers use two identical pumps, operating one at a constant speed and the second with a variable-speed drive to more continuously match delivery with demand. Such systems also require special analysis, and the pump manufacturer should be consulted.

13.15.5 Operating Pumps in Series

Directing the output of one pump to the inlet of a second pump allows the same capacity to be obtained at a total head equal to the sum of the ratings of the two pumps. This method permits operation against unusually high heads.

Figure 13.51 illustrates the operation of two pumps in series. Obviously, each pump carries the same flow rate Q_{total}. Pump 1 brings the fluid from the source, increases the pressure somewhat, and delivers the fluid at this higher pressure to pump 2. Pump 1 is operating against the head H_1 produced by the suction line losses and the initial increase in pressure. Pump 2 then takes the output from pump 1, further increases the pressure, and delivers the fluid to the final destination. The head on pump 2, H_2, is the difference between the total dynamic head *TDH* at the operating point for the combined pumps and H_1.

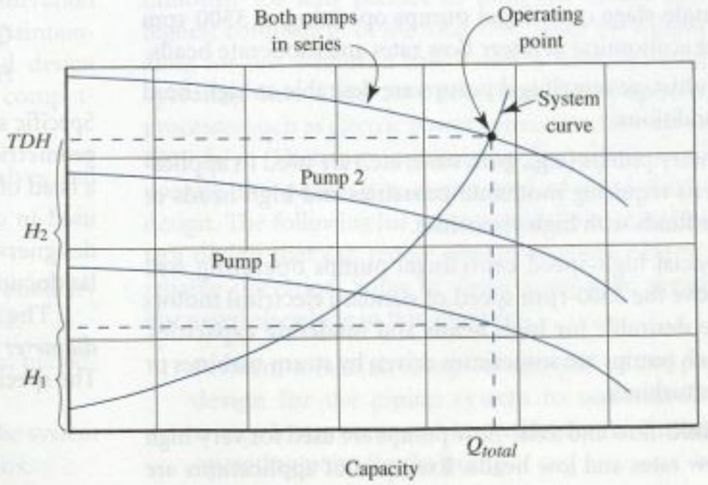
13.15.6 Multistage Pumps

A performance similar to that achieved by using pumps in series can be obtained by using multistage pumps. Two or more impellers are arranged in the same housing in such a way that the fluid flows successively from one to the next. Each stage increases the fluid pressure so that a high total head can be developed.

13.16 PUMP TYPE SELECTION AND SPECIFIC SPEED

Figure 13.52 shows one method for deciding what type of pump is suitable for a given service. Some general conclusions can be drawn from such a chart, but it should be emphasized that boundaries between zones are approximate.

FIGURE 13.51 Performance of two pumps operating in series.



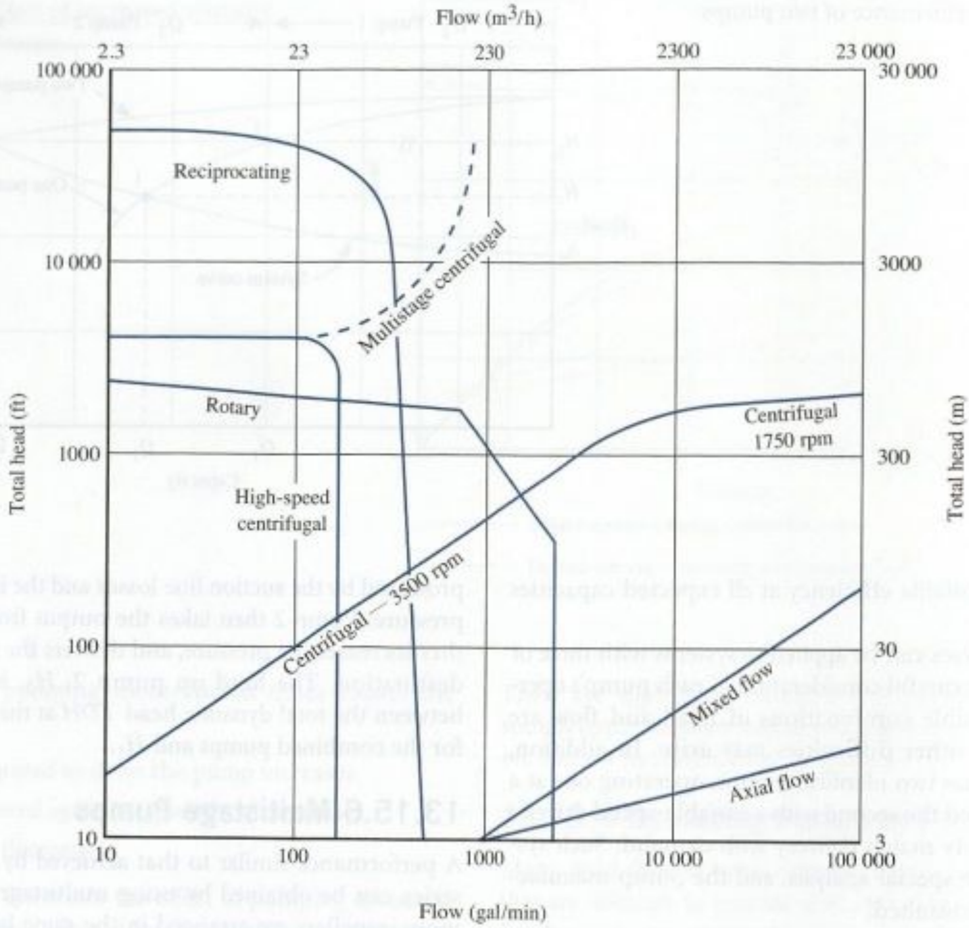


FIGURE 13.52 Pump selection chart.

Two or more types of pumps may give satisfactory service under the same conditions. Such factors as cost, physical size, suction conditions, and the type of fluid may dictate a particular choice. In general:

1. Reciprocating pumps are used for flow rates up to about 500 gal/min and from very low heads to as high as 50 000 ft.
2. Centrifugal pumps are used over a wide range of conditions, mostly in high-capacity, moderate-head applications.
3. Single-stage centrifugal pumps operating at 3500 rpm are economical at lower flow rates and moderate heads.
4. Multistage centrifugal pumps are desirable at high-head conditions.
5. Rotary pumps (e.g., gear, vane, etc.) are used in applications requiring moderate capacities and high heads or for fluids with high viscosities.
6. Special high-speed centrifugal pumps operating well above the 3500-rpm speed of standard electrical motors are desirable for high heads and moderate capacities. Such pumps are sometimes driven by steam turbines or gas turbines.
7. Mixed-flow and axial-flow pumps are used for very high flow rates and low heads. Examples of applications are flood control and removal of ground water from construction sites.

Another parameter that is useful in selecting the type of pump for a given application is the *specific speed*, defined as

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}} \qquad (13\text{–}17)$$

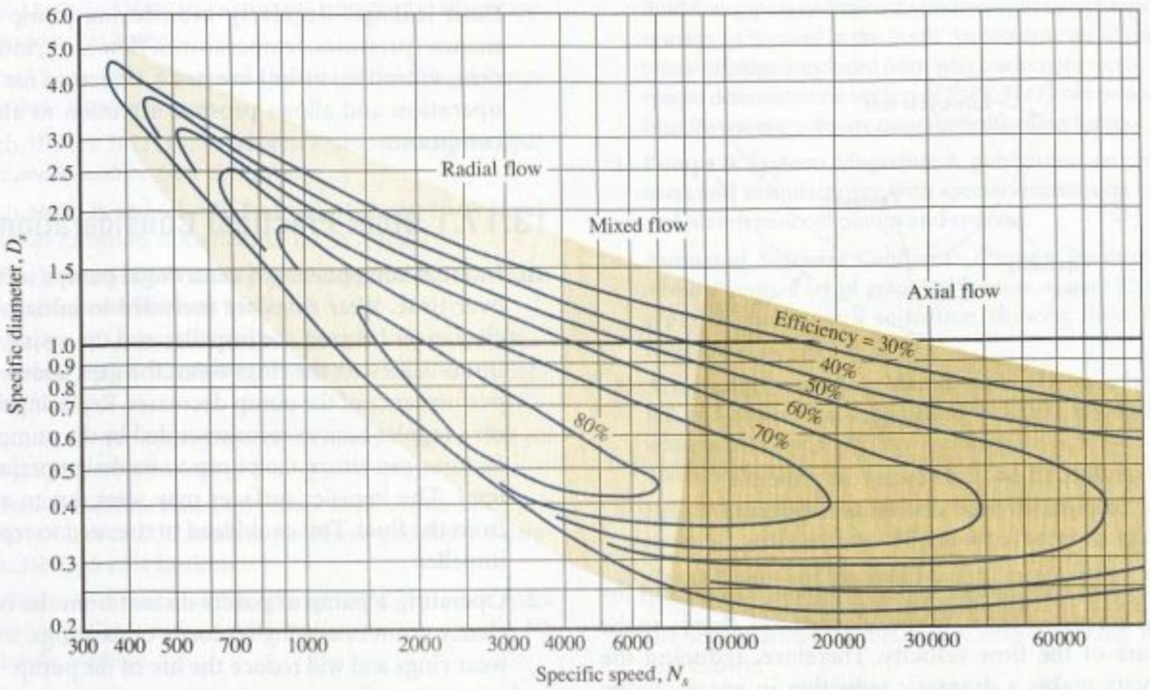
where

N = Rotational speed of the impeller (rpm)
Q = Flow rate through the pump (gal/min)
H = Total head on the pump (ft)

Specific speed can be thought of as the rotational speed of a geometrically similar impeller pumping 1.0 gal/min against a head of 1.0 ft (Reference 8). Different units are sometimes used in countries outside the United States, so the pump designer must determine what units were used in a particular document when making comparisons.

The specific speed is often combined with the *specific diameter* to produce a chart like that shown in Fig. 13.53. The specific diameter is

$$D_s = \frac{DH^{1/4}}{\sqrt{Q}} \qquad (13\text{–}18)$$



$$N_s = \frac{N\sqrt{Q}}{H^{3/4}} \qquad D_s = \frac{DH^{1/4}}{\sqrt{Q}}$$

N = Rev/min H = Head, ft
Q = Flow, U.S. gpm D = Diameter, in

FIGURE 13.53 Specific speed versus specific diameter for centrifugal pumps—An aid to pump selection. (Excerpted by special permission from *Chemical Engineering*, April 3, 1978. Copyright © 1978 by McGraw-Hill, Inc., New York)

where D is the impeller diameter in inches. The other terms were defined earlier for Equation 13-17.

From Fig. 13.53 we can see that radial-flow centrifugal pumps are recommended for specific speeds from about 400 to 4000. Mixed-flow pumps are used from 4000 to about 7000. Axial flow pumps are used from 7000 to over 60 000. See Fig. 13.12 for the shapes of the impeller types.

13.17 LIFE CYCLE COSTS FOR PUMPED FLUID SYSTEMS

The term *life cycle cost* (LCC) refers here to the consideration of all factors that make up the cost of acquiring, maintaining, and operating a pumped fluid system. Good design practice seeks to minimize LCC by quantifying and computing the sum of the following factors:

1. Initial cost to purchase the pump, piping, valves and other accessories, and controls.
2. Cost to install the system and put it into service.
3. Energy cost required to drive the pump and auxiliary components of the system over the expected life.
4. Operating costs related to managing the system including labor and supervision.
5. Maintenance and repair costs over the life of the system to keep the pump operating at design conditions.
6. Cost of lost production of a product during pump failures or when the pump is shut down for maintenance.
7. Environmental costs created by spilled fluids from the pump or related equipment.
8. Decommissioning costs at the end of the useful life of the pump including disposal of the pump and cleanup of the site.

More detail on each of these items and the larger context of LCC can be found in Reference 9.

Minimizing Energy Costs For pumps that operate continuously for long periods of time, the energy cost is the highest component of the total LCC. Even for a pump operating for just 8 hours a day, 5 days a week, the cumulative operating time is over 2000 h/yr. Pumps feeding continuous processes such as electric power generation may operate over 8000 h/yr. Therefore, minimizing the energy required to operate the pump is a major goal of good fluid system design. The following list summarizes the approaches to system design that can reduce energy cost and help to ensure reliable operation. Some of these items have already been discussed elsewhere in this chapter:

1. Perform a careful, complete analysis of the proposed design for the piping system to understand where energy losses occur and to predict accurately the design operating point for the pump.

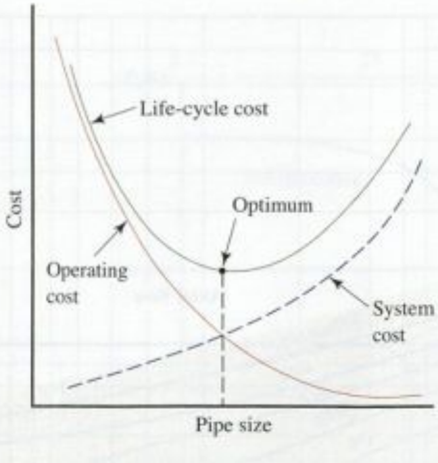


FIGURE 13.54 Life cycle cost principle for pumped fluid distribution systems.

2. Recognize that energy losses in piping, valves, and fittings are proportional to the velocity head, that is, to the square of the flow velocity. Therefore, reducing the velocity makes a dramatic reduction in energy losses and the total dynamic head required for the pump. A smaller, less expensive pump can then be used.
3. Use the largest practical size of pipe for both the suction and discharge lines of the system to keep the flow velocity at a minimum. Understand that larger pipes are more expensive than smaller pipes and they require larger, more expensive valves and fittings as well. However, the energy saving accumulated over the operating life of the system typically overcomes these higher costs. Figure 13.54 illustrates this point conceptually by comparing costs for the system with operating costs as a function of pipe size. Another practical consideration is the relationship between the pipe sizes and the sizes for the suction and discharge ports of the pump. Some designers recommend that the pipes be one size larger than the ports.
4. Carefully match the pump to the head and capacity requirements of the system to ensure that the pump operates at or near its BEP and to avoid using an oversized pump that will cause it to operate at a lower efficiency.
5. Use the most efficient pump for the application and operate the pump as close as possible to its BEP.
6. Use high-efficiency electric motors or other prime movers to drive the pump.
7. Consider the use of variable-speed drives (VSD) for the pumps to permit matching the pump delivery to the requirements of the process. See Section 13.15.2.
8. Consider two or more pumps operating in parallel for systems requiring widely varying flow rates. See Section 13.15.4.
9. Provide diligent maintenance of the pump and piping system to minimize the decrease in performance due to wear, build-up of corrosion on pipe surfaces, and fluid leakage. Regularly monitoring pump performance (pressures, temperatures, flow rates, motor current, vibration, noise) creates a signature for normal operation and allows prompt attention to abnormal conditions.

13.17.1 Other Practical Considerations

1. Internal components of centrifugal pumps suffer wear over time. Wear rings are included to initially set the clearances between the impeller and the casing to optimum values. As the rings wear, the gaps widen and the performance of the pump decreases. Replacing the rings on a regular basis as recommended by the pump manufacturer can return the pump to its design performance level. The impeller surfaces may wear due to abrasion from the fluid. This could lead to the need to replace the impeller.
2. Operating a pump at points distant from the best efficiency point causes higher loads on bearings, seals, and wear rings and will reduce the life of the pump.
3. Rigid support for the pump and its drive system is critical to satisfactory operation and long life.
4. Careful alignment of the drive motor with the pump is essential to avoid excessive deflection of the pump's shaft that can cause early failure. Follow the pump manufacturer's recommendations and check alignment periodically.
5. Assure that the flow from the suction line to the pump inlet is smooth with no vortices or swirling. Some designers recommend a minimum of 10 diameters of straight pipe ($10 \times D$) between any valve or fitting and the pump inlet. However, if a reducer is required, it should be installed right at the pump.
6. Support piping and valves independently from the pump and do not allow significant pipe loads to be transferred to the pump casing. High loads tend to cause extra loads on bearings and shaft deflection that may change the clearance between the impeller and the casing.
7. Use clean oil, grease, or other lubricants for the pump bearings.
8. Do not allow the pump to operate dry or with entrained air in the fluid being pumped. This requires careful design of the intake to the suction line and the tank, sump, or reservoir from which the fluid is drawn.

REFERENCES

Note: See also the extensive list of references for Chapter 11, many of which treat the subject of pumps and pumped systems.

1. ASTM International. 2008. *ASTM D323-08 Standard Test Method for Vapor Pressure of Petroleum Products (Reid Method)*. DOI: 10.1520/D0323-08. West Conshohocken, PA: Author.
2. ———. 2012. *ASTM D4953-06(2012). Standard Test Method for Vapor Pressure of Gasoline and Gasoline-Oxygenate Blends*